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AN EVALUATION OF THE EFFECTS OF LUBRICANTS ON THE THERMODYNAMIC PROPERTIES AND PERFORMANCE OF REFRIGERANT MIXTURES IN REFRIGERATION AND AIR CONDITIONING CYCLES.

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ABSTRACT

A thermodynamic model based on a Wilson-MHV-2 methodology has been developed to predict the thermodynamic properties of quaternary mixtures comprising the refrigerants R-32, R-125 and R-134a and an ISO32 mixed-acid polyol ester lubricant.

The interaction parameters for this model have been derived by fitting to experimental VLE data for binary mixtures. The capability of the Wilson-MHV-2 model to predict the VLE behaviour of the quaternary lubricant-refrigerant mixtures has been examined by comparison with experimental VLE data measured directly for a specific mixture composition. The model has been used to examine the effect of the presence of a polyol ester oil in perturbing the thermodynamic and performance properties of binary and ternary mixtures of HFC-32, HFC-125 and HFC-134a at different points around the refrigeration cycle. Under most normal system operating conditions, the effects of oil are expected to be small but may increase significantly for those points in the cycle that are operating close to the dew point line of the oil-free refrigerant.

INTRODUCTION

Although they are by no means a new phenomenon, the development of a range of zeotropic refrigerants as alternatives to R-22 and R-502 has brought into industry focus the issues surrounding the utilisation of such refrigerants. Some of the main industry concerns around the use of zeotropic refrigerants have been in the areas of;

- composition shifts in systems
- system leakage & top-up effects
- refrigerant glide utilisation
- performance assessment relative to conventional refrigerants
- bulk storage & handling

There have been a number of publications dealing with these issues, most of which are now either well understood or in the process of being defined (Corr *et al* 1994, Ferrari *et al* 1994, Kruse *et al* 1995).

Whilst many of these aspects can be readily addressed by using models based on refrigerant mixture thermophysical properties alone, the behaviour and performance of refrigerants in systems is known to be affected by the presence of significant quantities of lubricants. In effect, the functional mixtures to be found around most refrigeration systems are zeotropic mixtures of refrigerant fluids with the lubricating oil. Although many researchers are measuring and predicting the thermodynamic properties of the new HFC refrigerant blends, there has been relatively little work on including the lubricant in these model descriptions or in theoretical refrigeration cycles.

The behaviour of many refrigerant-lubricant mixtures have now been studied, primarily from the point of view of lubricant return and the properties (viscosity and surface tension) of the compressor sump contents, using VLE and solubility data. (Henderson *et al* 1994) have utilised such data to rationalise the effect of lubricant on system evaporator and condenser behaviour, (Corr *et al* 1994) have used differential refrigerant solubility in sump oil as part of a model dealing with composition shifts of HFC blends in functioning systems. Whilst this approach is valuable, it does have limitations when compared to an approach that utilises an equation of state. The development of an equation of state model that can be used to predict the properties of mixtures across a wide range of both refrigerant mixture and refrigerant-lubricant mixture compositions and conditions offers the potential to be a much more flexible and useful approach. The principal advantage of using the latter is that state properties e.g. enthalpy and entropy, which are central to cycle performance,

can be calculated directly for the oil-refrigerant mixture at any system condition. In addition, such a model would bring the ability to predict the vapour liquid equilibria (VLE) properties and vapour phase properties of zeotropic refrigerant-lubricant mixtures within the temperature and pressures ranges relevant to the refrigeration and air-conditioning industries. This work focuses on using such an equation of state to predict the thermodynamic properties of refrigerant-lubricant systems and, in particular, applying it to evaluate the perturbation effect of the presence of oil on the refrigeration cycle properties.

EXPERIMENTAL

A schematic diagram of the experimental apparatus for binary refrigerant-lubricant VLE data is given in figure 1.

MODEL DESCRIPTION

Previous work (Morrison *et al*, 1995) has shown that the MHV-2 approach (Huron 1979, Michelsen 1990a, b, Dahl 1990) is able to predict the properties of ternary mixtures comprising R-32, R-125 and R-134a to less than 1% difference in pressure over the saturation temperature range of interest (-40 C to 60 C). An attractive feature of this approach is that since it is based on a cubic equation of state it has few model parameters and can be developed using a limited quantity of binary mixture experimental data. It therefore offers a compromise of moderate accuracy from limited experimental data rather than high accuracy from a more complex model that would require large amounts of experimental data. In the current work the MHV-2 approach has been extended to include the lubricant, which is very much less volatile than the three refrigerants, as a fourth component. As a test of applicability, the MHV-2 approach has been used to predict the VLE properties of this quaternary mixture for comparison with direct experimental measurements (figure 2). Over the temperature range examined, the agreement between model and experimental pressures is better than 2.5%, particularly promising given the difficulties in measuring accurate refrigerant-lubricant binary VLE data, especially at low temperatures and pressures.

CYCLE PERFORMANCE CALCULATIONS

The steady state quantity of lubricant present in a refrigeration system varies widely from around 0.1% by weight of refrigerant for many small hermetic compressors to around 10% or so in an automotive air conditioning system. In addition, the cycling behaviour of the compressor can lead to periods of abnormally high lubricant content in the system. In the following examples, a lubricant content of 2% was chosen as a representative value for many commercial installations. It should be noted that although the effects described have been exemplified using zeotropic HFC refrigerant blends, similar effects are obtained with azeotropic blends or even pure fluids. It is the presence of the relatively involatile lubricant and its interaction with the refrigerants that is responsible for the estimated effects described below.

Model Evaporation and Condensation Process

Consider a simple model evaporation process that occurs at constant pressure and with vapour-liquid equilibrium along the length of the evaporator. Also, for simplicity, it is assumed in this particular example that at the evaporator inlet the refrigerant is at the bubble point. This process is represented in the pressure enthalpy diagram in figure 3 by points 1 to 2. Figure 5 compares the calculated temperature profiles for R-407A (20/40/40 R-32/R-125/R-134a) alone and with 2% oil as a function of mixture quality. The pressure was taken as 1.3 bar for all of these calculations. It can be seen that the presence of the oil does not affect the calculated temperatures significantly until the mixture quality exceeds 95%.

Normally, evaporation and condensation of zeotropic mixtures are described with reference to the dew and bubble points of the mixtures. With boiling points in excess of 300C, addition of the lubricant significantly increases the dew point of any refrigerant mixture (figure 4). The refrigerant-lubricant mixture bubble point is, as expected, dominated by the refrigerant bubble point properties. The complete refrigerant-lubricant mixture evaporation process effectively occurs in two stages, the first being essentially the evaporation of the refrigerant component and the second being the evaporation of the oil. With the temperatures normally encountered in refrigeration and a/c applications, the second part

of the evaporative process does not take place to any significant extent. In order to evaluate the effect of the oil on the evaporative process in a sensible manner, it is more informative to define the outlet temperatures for the evaporator from the refrigerant in the absence of oil. This could either be the dew point of the refrigerant blend (point 2 figure 3) or some superheated temperature (point 2' figure 3).

With the inlet and outlet conditions (P,T) defined in this way for the refrigerant-lubricant blend, the effect of adding oil can now be considered by performing thermodynamic property calculations by simply including the lubricant as the fourth fluid component in the MHV-2 based model. Figure 5 shows a comparison of the calculated refrigerant mass fraction in the liquid phase for R-407A without oil compared to that with 2% oil as a function of temperature. The outlet condition for the evaporation process is the dew point of R-407A (without oil). It is clear from figure 5 that for all of the temperatures below -35 C, the oil has very little influence on the evaporative process. However, from -35 C to -33 C (R-407A dew point) some residual refrigerant remains in the liquid phase in the case when the oil is present. This is a manifestation of the refrigerant solubility in the oil. Figure 6 shows the effect of the presence of 2% oil on R-407A between the inlet (1) and superheated outlet temperatures (2'). The difference in the calculated enthalpy, the residual liquid mass fraction and the residual liquid refrigerant mass fraction are displayed as percentage values. The first shows a pronounced minimum and the other two a maximum, close to the dew point of R-407A. As the temperature is increased above the conventional (refrigerant-only) dew point, the solubility of refrigerant in the lubricant gradually diminishes and more refrigerant goes into the vapour phase. This has the effect of diminishing the perturbation effect of the oil from a calculated maximum of 4% for the enthalpy to around 1.4%.

The effect of the presence of the oil on the condensation process has been evaluated in the same way as described above for the evaporation process. For example, figure 7 compares the temperatures for R-407A with and without 2% added oil and plotted as a function of total liquid mass fraction. The magnitude of the enthalpy change due to the presence of the oil is shown in figure 8. As with the evaporation process, figure 8 compares the properties across temperatures that range from the bubble point of R-407A at 20 bar to 20 K above the conventional dew point. The maximum perturbation is at the dew point of R-407A i.e. at or close to the condenser inlet, and is slightly larger than that observed at the lower pressures (1.3 bar) relevant to evaporation (5.6% c.f. 4.0%).

Isentropic Compression Process

The effect of oil on the heat of compression has been studied by calculating the enthalpy of isentropic compression of the refrigerant-lubricant mixtures from the suction line pressure to a chosen discharge pressure. Two different compressor inlet temperatures were considered as examples, each at suction line pressure of 1.3 bara. The first corresponds to the calculated dew point temperature of R-407A i.e. -33 C (point 2 in figure 3), the second corresponds to 20 K above the conventional dew point temperature i.e. -13 C. For each of the two compressor inlet conditions, four discharge conditions were studied; 15, 20, 25 and 30 bara. The calculated heats of compression for R-407A and R-407A with 2% oil are shown in figure 9. This clearly shows that the differences in calculated heat of compression are significantly greater for compressor inlet conditions at or near the dew point of the refrigerant blend than those further into the superheat region.

Evaporator Operating Condition Effects

The refrigeration effect is represented by the enthalpy difference between point 1 and point 2 in figure 3 (for simplicity, it is assumed that 100% liquid is present at the evaporator inlet i.e. that the expansion device flash enthalpy contributes to the refrigeration effect). In this example, the refrigerant R-407C (23/25/52 R-32/R-125/R-134a) has been used to illustrate the effect of evaporator pressure on the refrigeration effect. Figure 10 shows the calculated refrigeration effect for R-407C with 0.5%, 1% and 2% oil at three evaporator pressures, 1, 3 and 5 bara. In the absence of any superheat, 1% oil reduces the refrigeration effect to approximately 95% of that calculated for R-407C alone. The magnitude of this influence increases with increasing evaporator pressure, a manifestation of the increased solubility of the refrigerants in the lubricant at higher pressure. The influence of the oil on the refrigeration effect increases roughly in proportion to the mass of oil present. As with figure 7, the magnitude of this reduction decreases with the addition of superheat beyond the conventional dew point.

Energy Efficiency

The effect of the presence of oil on the calculated energy efficiency has been studied at eight different cycle conditions. Using figure 3, the first four sets of conditions are represented by inlet evaporator conditions at point 1, outlet at point 2 the dew point for R-407A and then compression to one of the four discharge pressures, represented by points 3(a), 3(b), 3(c) and 3(d). The latter correspond to 15, 20, 25 or 30 bara. The second four sets of conditions are represented by inlet evaporator conditions at point 1, outlet at point 2' 20 K above the bubble point temperature of R-407A and then compression to one of the four discharge pressures, represented by points 3'(a), 3'(b), 3'(c) and 3'(d). The results of the COP calculations for R-407A with 2% oil are shown in figure 11. The difference in the COPs relative to R-407A have been calculated. It is clear from figure 11 that the extent to which the presence of oil influences the calculated COP is dependent on the amount of superheating prior to the compression process. With 2% oil circulating around the system the maximum reduction in COP is estimated to be around 10% from the value for the pure refrigerant alone. This effect diminishes rapidly with the addition of superheat, being minimal by the 20K superheat level.

CONCLUSIONS

Calculation results for simple model evaporation, condensation and isentropic compression processes have shown that the presence of oil at levels of 1-2% can affect the calculated refrigeration effect, heat of compression and COP. Under most normal system operating conditions, the effects of oil are expected to be small but may become significant for those points in the cycle that are operating close to the dew point line of the oil-free refrigerant. Although the work presented here has focused on refrigerants from the R-400 series, it should be emphasised that the findings are equally applicable to other systems including single component and azeotropic refrigerants. These findings are in good agreement with what has been observed experimentally (McMullen *et al* 1988) where pronounced reductions in refrigeration effect and COP are observed at low levels of superheat with R12 and mineral oil. These calculation results emphasise the importance of including lubricant effects in the development of detailed cycle performance models. The work described in this paper shows that the approach of using an equation of state based model to describe lubricant-refrigerant systems is promising and should be developed further for use in theoretical cycle performance analysis.

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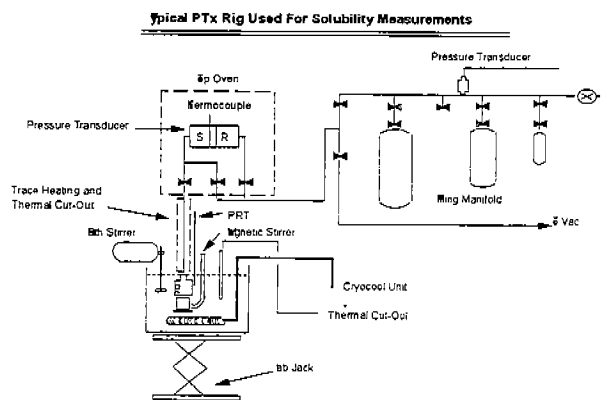


Figure 1 Schematic diagram of apparatus used for experimental VLE measurements

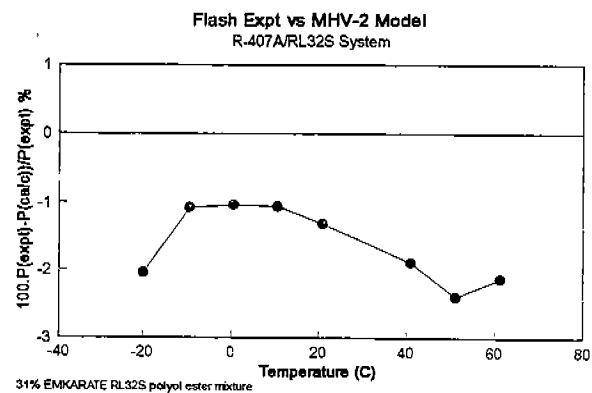


Figure 2 Calculated versus experimental vapour pressures for quaternary R-32, R-125, R-134a, POE mixtures

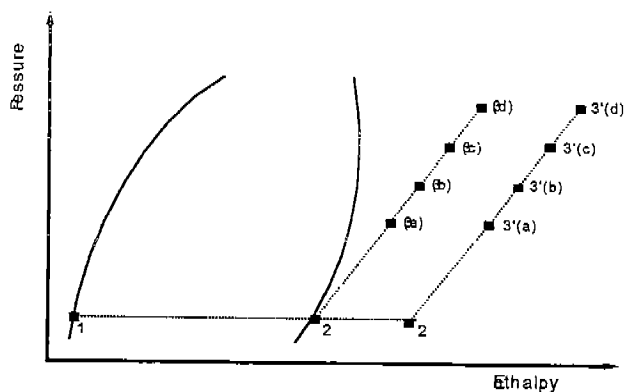


Figure 3 Schematic pressure-enthalpy diagram

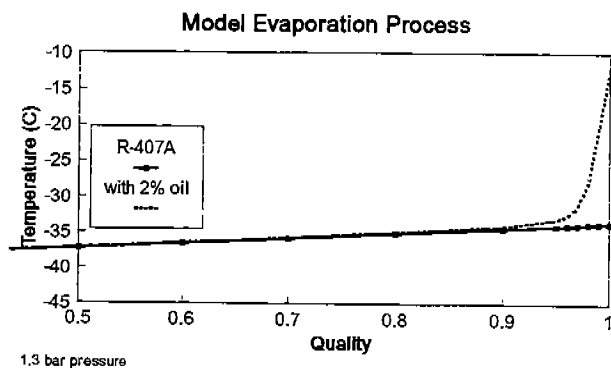


Figure 4 Comparison of calculated temperatures for R-407A and R-407A with 2% POE oil

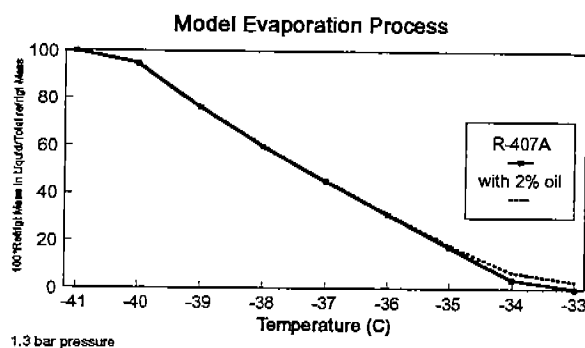


Figure 5 Comparison of calculated refrigerant mass fraction in the liquid phase as a function of temperature

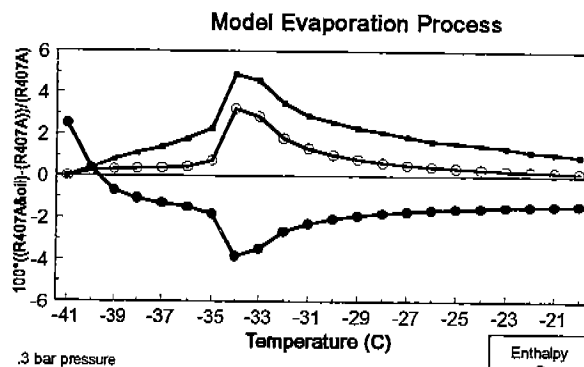


Figure 6 Calculated % differences in Enthalpy, Residual Liquid Mass Fraction and Refrigerant (only) Mass Fraction for R-407A with 2% POE oil.

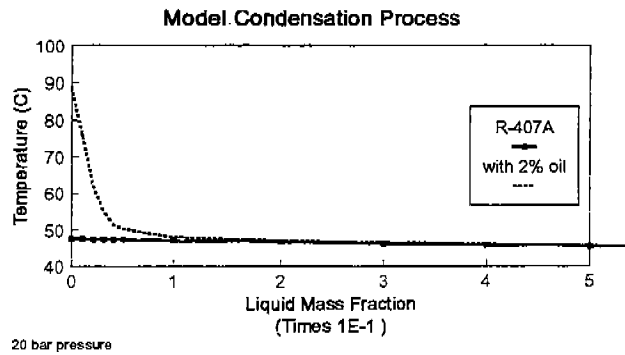


Figure 7 Comparison of calculated temperatures for R-407A and R-407A with 2% POE oil

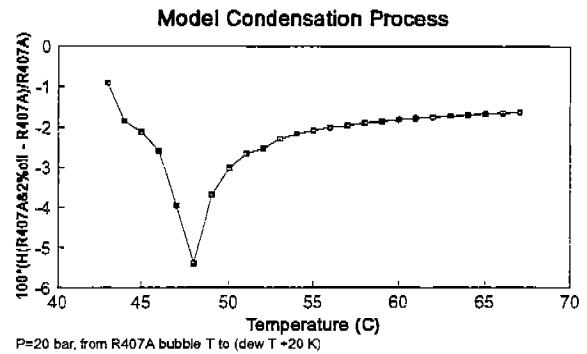


Figure 8 Calculated %Enthalpy difference for R-407A with and without 2% POE oil.

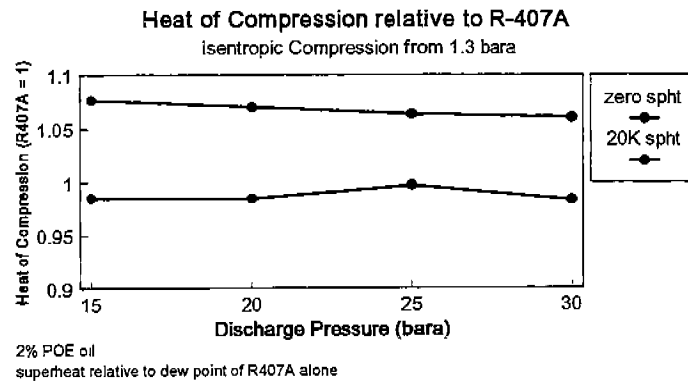


Figure 9 Calculated effects of oil and superheat on the enthalpy of isentropic compression

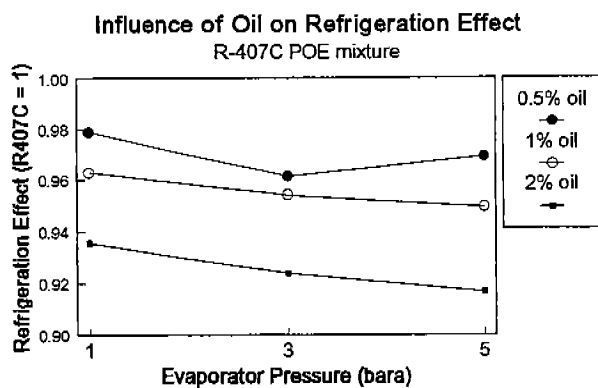


Figure 10 Influence of evaporator conditions on the effects of oil on the evaporation process

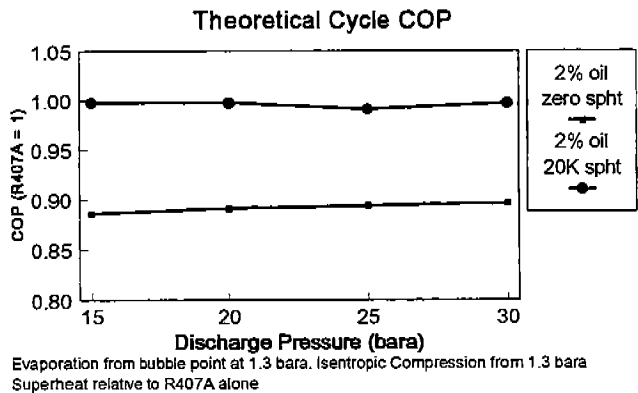


Figure 11 Effects of oil and superheat on the calculated theoretical cycle COP